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Static and Dynamic Analysis on R410A Scroll Compressor Components

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ABSTRACT

This study investigated the static and dynamic phenomena on R410A scroll compressor components by using finite element analysis (FEA) with ITRI self-developed scroll type compressor software (ITRI_STC software). The simulation results of the coefficient of performance, cooling capacity by tuning the design parameters integrated scroll geometric parameters with compressed discharge procedure and dynamic equilibrium were presented by ITRI_STC software. The analyzed parts were fixed scroll, orbiting scroll, driving crank, and assemble part. The thermal and structural loadings were from ITRI_STC software. Those static and dynamic results were obtained by proper choices of boundary conditions. The temperature distribution, deformation, stress and modal shape were demonstrated in this paper. The final results were verified with the experiment pictures. The proposed procedures provide the failure problems of the compressor in advance. That information has provided designers to make some modifications for the weaker structural parts. Without making several real prototypes, it saves a lot of time and cost in the development of new compressor.

1. INTRODUCTION

Lin *et al.* (2005) observed temperature distribution and thermal deformation on the fixed and orbiting scroll for R22 refrigerant. The investigation for R410A scroll compressor was addressed this paper. R410A is a clean, low toxicity, non-flammable, zero ozone depletion potential and high cooling capacity refrigerant. The saturated working pressure of R410A refrigerant is approximately 1.6 times that of R22 refrigerant (Honeywell), so the mechanics analysis of those major component parts in the scroll compressor is important. Lots of papers showed the design logic from the thermodynamic or geometry view. (Chen *et al.* 2002; Blunier *et al.* 2009; Duprez *et al.* 2010; Cuevas *et al.* 2010) The design of scroll compressor should consider not only thermodynamic or geometry but also mechanics. For instance, where is the best location for the bearing? What resonant speed should we avoid? Where is the weakest part of components? Those questions make authors do the further research on scroll compressor. This proposed procedure take further investigations on several physic parameters such as temperature distribution, total deformation, stress distribution and modal shape of scroll compressor. There are few literatures addressed the analysis for scroll compressor from mechanics view, especially the mode shape issues on assemble components. The paper is organized as follows. Section 2 presents ITRI_STC software overview. Section 3 presents finite element analysis. Section 4 presents design procedure and section 5 shows static and dynamic simulation results.

2. ITRI_STC SOFTWARE OVERVIEW

ITRI self-developed scroll compressor software integrated scroll geometric parameters, compressed exhaust processes and dynamic equilibrium, provided simulation results such as the COP, air-conditioning capacity, etc. through the design parameters changed. The input design parameters were: common scroll, orbiting and fixed scroll, Oldham ring, eccentric shaft, bearings, other compressor components, friction coefficient, calories meter parameters, motor speed and efficiency, the choice of refrigerant and lubricant, and compressor, discharge angle, by-pass, valve, refrigerant spray, motor input architecture. This software could compute multiple parameters simultaneously. The output parameters were: power balance and power consumption, journal bearings maximum estimation, performance parameters, motor output power, cold room capacity, geometric restrictions parameters and the power consumption. The design engineer could modify reasonable design parameters according the simulated results for the compressor.

2.1 The Geometric Model of Scroll Compressor

When orbiting and fixed scroll are meshed with each other, it is only the line mesh rather than the surface mesh. The distance between two base circles doesn't vary with the main shaft angle change during mesh procedure. The line of 2 scroll meshing point is parallel to the 2 base circles connection and tangent to the base circle. The fixed scroll shape can get from the mirror image of the rotated 180 degrees orbiting scroll shape. The base circle radius is a . The spiral pitch is $P = 2\pi a$. The spiral initial angle is α . The launch angle of the spiral is ϕ . The scroll wall thickness is $t = 2a\alpha$. The height of scroll is h . The scroll number is N . The spiral opening angle is $\alpha = 2\pi N + \pi/2$. The fixed and orbiting scrolls base circle center distance (orbiting radius) is $r = P/2 - t$. The simple mathematical formulas are as follows:

$$\text{The outer spiral of scroll: } x_0 = r[\cos\phi + (\phi + \alpha)\sin\phi]; y_0 = r[\sin\phi - (\phi + \alpha)\cos\phi] \quad (1)$$

$$\text{The inner spiral of scroll: } x_i = r[\cos\phi + (\phi - \alpha)\sin\phi]; y_i = r[\sin\phi - (\phi - \alpha)\cos\phi] \quad (2)$$

2.2 Scroll Compressor Thermodynamic Model

This model considers the impact of the compressor work process such as the suction superheat, suction compression, leakage, discharge compression factors. The scroll compressor thermodynamic model was established by mass and energy conservation basis. Due to many factors affect the scroll compressor, we have the following assumptions to simulate the work process:

(1) Gas is an ideal gas; specific heat is the fixed value; oil droplet is no phase transition; gas is incompressible.

(2) Oil and gas homogeneous mixture within control volume at any time which means medium state parameters at any point is the same. The influences from outside are passed to media within control volume immediately. According to variable mass thermodynamics viewpoint, the flow phenomena through the aperture and leakage gap are mass exchange between control volume and the outside, integrated the thermodynamic model of the suction, compression, discharge stage, establish the basic thermodynamics equation for working chamber:

$$\text{Conservation of energy: } \frac{\partial T}{\partial \theta} = \frac{1}{mC_v} \left\{ -T \left(\frac{\partial P}{\partial T} \right)_v \left[\frac{\partial V}{\partial \theta} - \frac{v}{\omega} \right] - \sum \frac{\dot{m}_m}{\omega} (h - h_m) + \frac{\dot{Q}}{\omega} \right\} \quad (3)$$

$$\text{Conservation of mass: } \frac{\partial m}{\partial \theta} = \sum \frac{\dot{m}_m}{\omega} - \sum \frac{\dot{m}_{out}}{\omega} \quad (4)$$

T is the refrigerant temperature. θ is scroll lunch angle. m is the refrigerant mass. C_v is the specific heat. P is the refrigerant pressure. V is control volume. v is the specific volume. ω is the angular velocity of the compressor shaft. \dot{m} is the mass flow rate. h is the enthalpy for the refrigerant within the control volume. h_m is the enthalpy for the refrigerant entering the control volume. \dot{Q} is the heat rate for the refrigerant entering the control volume.

2.3 Scroll Compressor Dynamic Model

The upper and lower counterweight mass can be computed by the mass and location of all components through the resultant forces of shaft and the torque balance. From the lower counterweight $mLCW$, we compute torque balance of inertia force $mLCWrLCW$ based on upper counterweight $mUCW$. Then we compute the torsion inertia torque balance $mUCWrUCW$ for upper counterweight. According to the given dynamic radius of mass center for lower and upper counterweight $rcUCW$ & $rcLCW$, we can get the mass of counterweight. The prefix m means mass. The prefix r means radius. The prefix y means the location along y direction. The rest are the name of all other parts.

$$\begin{aligned} mLCWrLCW = & -\{mScroll \times r \times (yScroll - yUCW) \\ & + mCrank \times r \times (yCrank - yUCW) \\ & + mOldham \times r \times (yOldham - yUCW) \\ & + mShaftboss \times r \times (yShaftboss - yUCW)\} / (yLCW - yUCW) \end{aligned} \quad (5)$$

$$\begin{aligned} mUCWrUCW = & -\{mScroll \times r + mCrank \times r + mOldham \times r \\ & + mShaftboss \times r + mLCWrLCW\} \end{aligned} \quad (6)$$

$$\begin{aligned} mLCW &= mLCWrLCW / rcLCW \\ mUCW &= mUCWrUCW / rcUCW \end{aligned} \quad (7)$$

The radial force F_r , tangential force F_θ , the axial force F_a and the centrifugal force F_c from the thermodynamics can compute the forces for upper lower bearing, crank shaft and the upper and lower counterweight.

2.4 The Outputs from STC software

Coefficient of performance (COP) and some size restrictions:

Performance parameter is $COP = \dot{Q}_c / P_{motor}$. (8) \dot{Q}_c is the cold room capacity. P_{motor} is the total power consumption. They are all relative to the design size and speed.

$$\text{Scroll slenderness ratio is } G_w = h_e / t \quad (9)$$

$$\text{Tool slenderness ratio is } G_w = h_e / (p_t - t) \leq 2.5 \quad (10)$$

$$\text{The smallest diameter for moving around is: } D_{ob_min} = 2\sqrt{x_{ob_o}^2 + y_{ob_o}^2} = (p_t / \pi) \sqrt{1 + \phi_r^2} \leq D_{0_max} \quad (11)$$

h_e is the height of scroll. t is the thickness of the scroll. The spiral angle for scroll spiral is ϕ . p_t is the pitch.

3. FINITE ELEMENT ANALYSIS

3.1 Software Introduction

The software packages, SolidWorks and ANSYS, were used in this analysis. SolidWorks and ANSYS are software package for 3D CAD design model and finite element analysis respectively. ITRI_STC software provided the thermal and structural loading estimations. Those estimations were viewed as the boundary conditions for ANSYS simulation. (Madenci *et al.* 2005; Stolarski *et al.* 2007)

3.2. Material Properties

The radial compliant mechanism, bush, and shaft are made of structural steel. The rotor is made of aluminum alloy. The scroll, upper and lower counterweights are made of stainless steel. The effective materials shown here were modified to protect the intellectual property rights

3.3 Scroll Dimensions

When the orbiting angle is 130° , there is maximum compression ratio. The maximum compression ratio was defined as the ratio of the maximum suction volume and the maximum discharge volume. The zero degree was defined as the mesh point for maximum suction volume. The clockwise direction was defined as positive direction. The effective angle shown here were modified to protect the intellectual property rights

4. DESIGN PROCEDURE

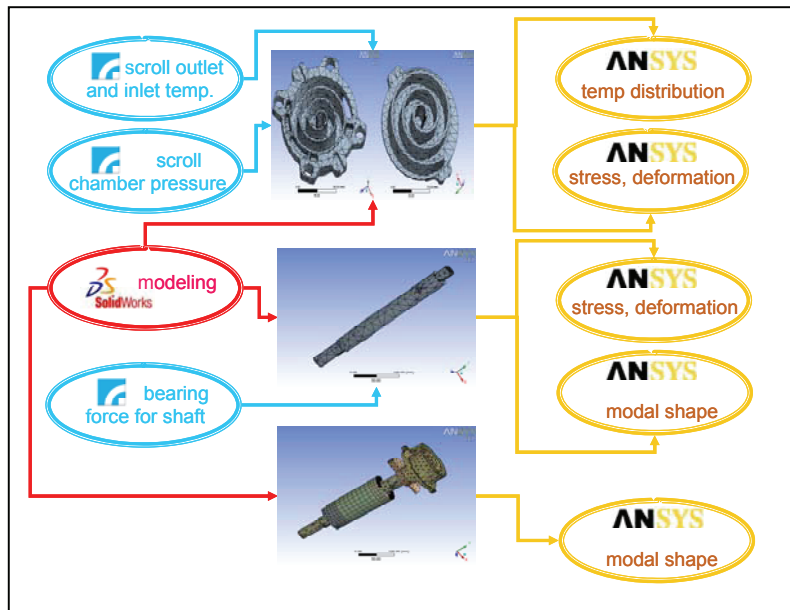


Figure 1: The analyzed procedure by ANSYS for scroll type compressor

ITRI_STC, SolidWorks, and ANSYS software were used in this study. The procedure in this study was following: First, ITRI_STC provided the chamber pressure and bearing force. Those parameters substituted into the SolidWorks based 3D model in ANSYS. By the proper boundary conditions choices, the temperature distribution, deformation, stress, and modal shape were gained. This paper demonstrated the fixed and orbiting scrolls temperature distributions; the fixed, orbiting scrolls and shaft deformations; the fixed, orbiting scrolls and shaft stresses; the shaft and assemble components modal shapes.

5. STATIC AND DYNAMIC SIMULATION RESULTS

ANSYS is a good tool for the thermal-structural coupled analysis. The boundary conditions are divided into three parts: temperature for compressor inlet and outlet, chamber pressure for working fluid and physical constrains. The temperatures are measured from experiment. The chamber pressures are calculated from ITRI_STC software. The physical constrains are set from the operating action. Those conditions are shown as follows.

The thermal boundary conditions for fixed scroll are: a) the suction temperature is 35°C , b) the discharge temperature is 105°C , and c) the surrounding temperature is 69°C .

The structural boundary conditions for fixed scroll are: a) the high-side and intermediate pressure estimated from ITRI_STC software are 5 MPa, b) the low-side pressure is 1.5 MPa, and c) the back pressure is 2 MPa.

Constrains for fixed scroll are: a) the four circle holes are frictionless support, and b) the two key ways are fixed.

The thermal boundary conditions for fixed scroll are: a) the suction temperature is 35°C , b) the discharge temperature is 105°C , and c) the surrounding temperature is 69°C .

The structural boundary conditions for fixed scroll are: a) the high-side and intermediate pressure estimated from ITRI_STC software are 5 MPa, b) the low-side pressure is 1.5 MPa, and c) the back pressure is 2 MPa.

Constraints for orbiting scroll are: a) the concentric circle part for thrust bearing surface is frictionless support, and b) the two key ways are frictionless support

The effective numbers shown here were modified to protect the intellectual property rights.

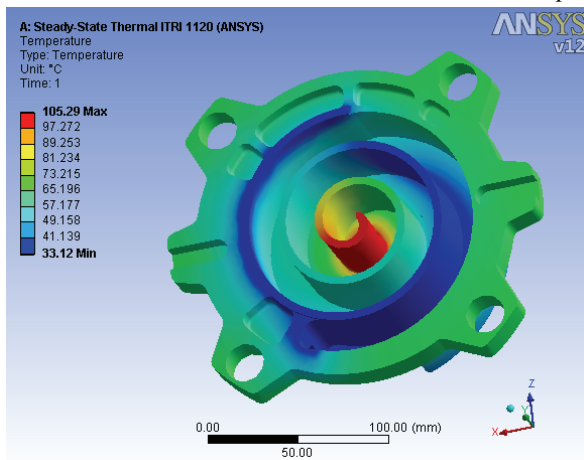


Figure 2 the temperature distribution for fixed scroll

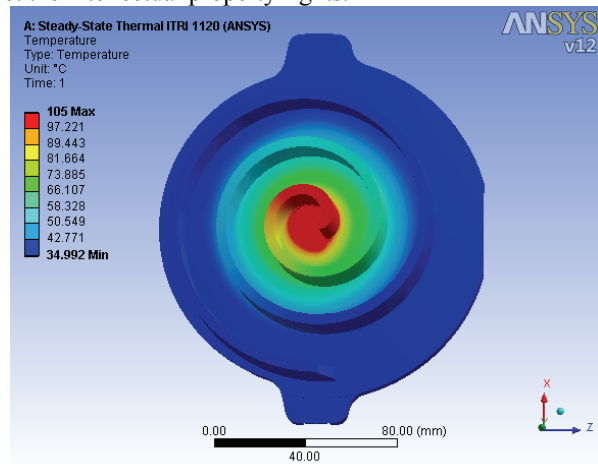


Figure 3 the temperature distribution for orbiting scroll

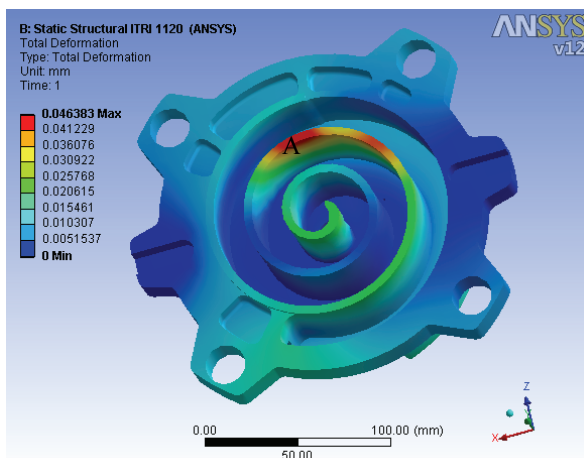


Figure 4 the total deformation for fixed scroll

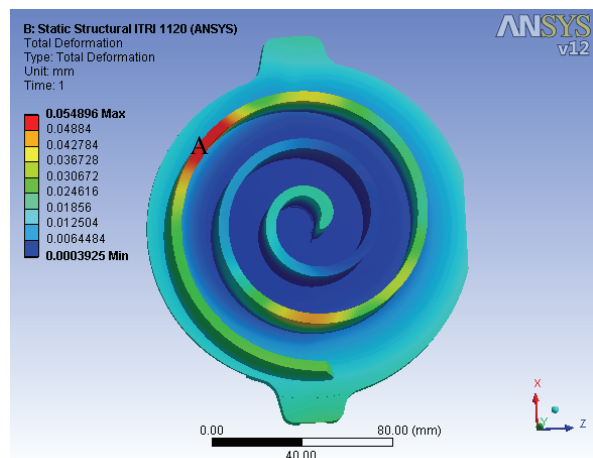


Figure 5 the total deformation for orbiting scroll

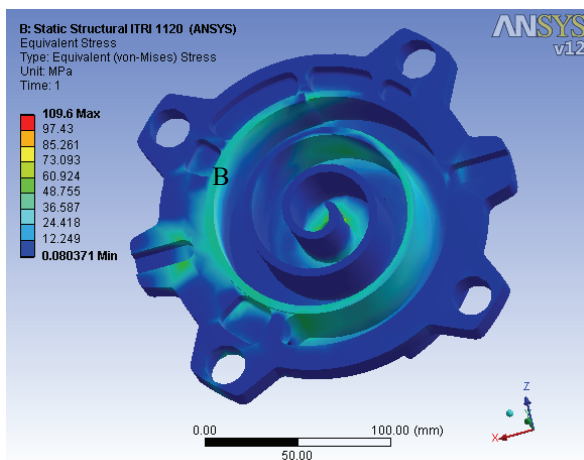


Figure 6 the stress distribution for fixed scroll

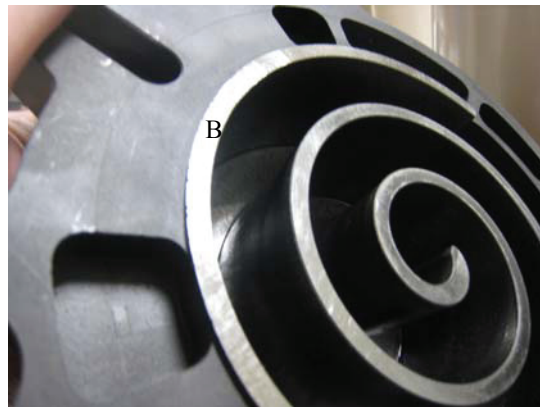


Figure 7 the polished area for fixed scroll

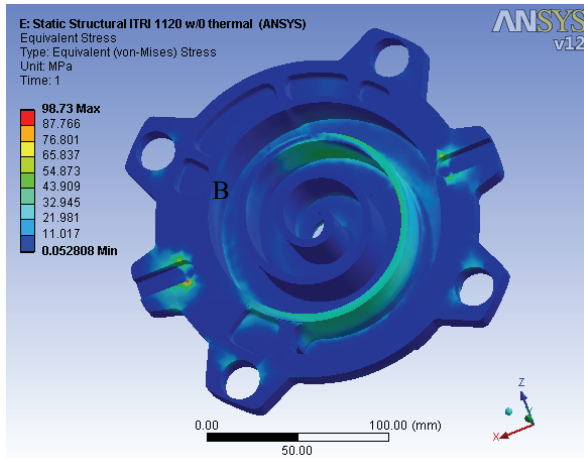


Figure 8 the stress distribution for fixed scroll without temperature effect

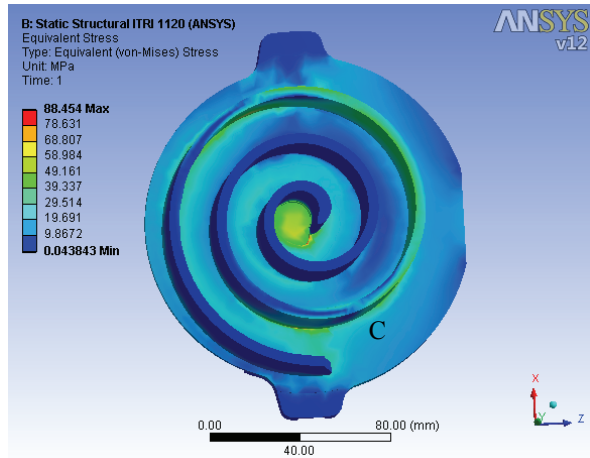


Figure 9 the stress distribution for orbiting scroll

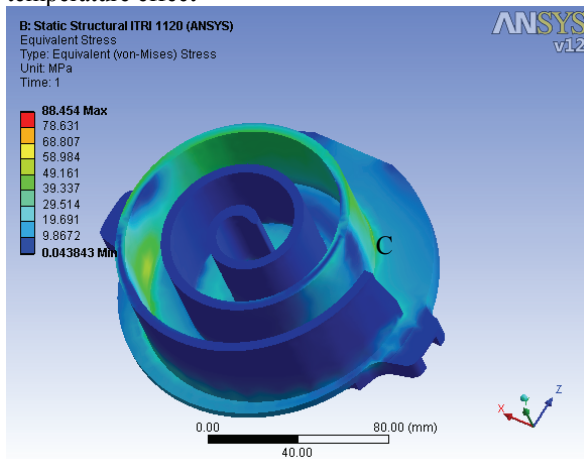


Figure 10 the stress distribution for orbiting scroll



Figure 11 the polished area for orbiting scroll

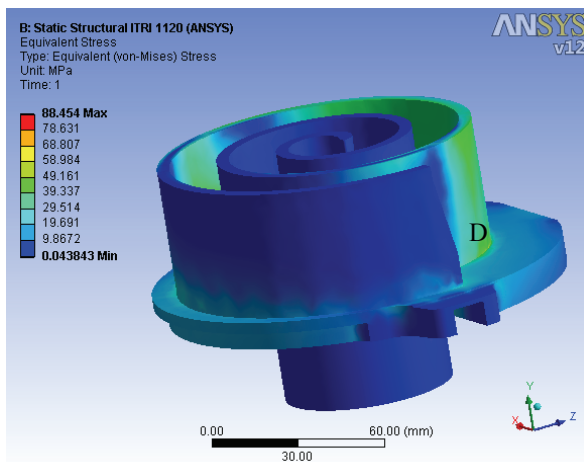


Figure 12 the stress distribution for orbiting scroll



Figure 13 the stress distribution for orbiting scroll

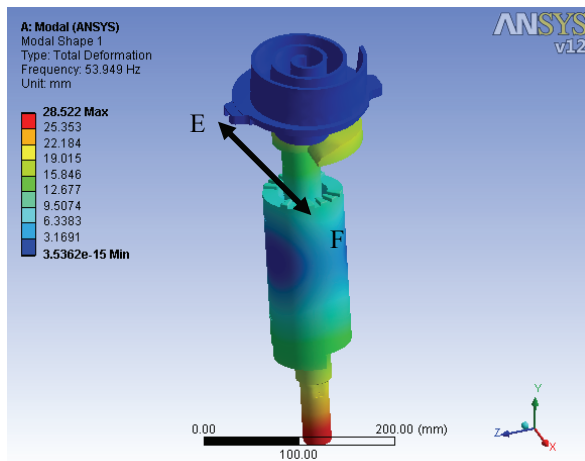


Figure 14 the 1st modal shape for assemble parts

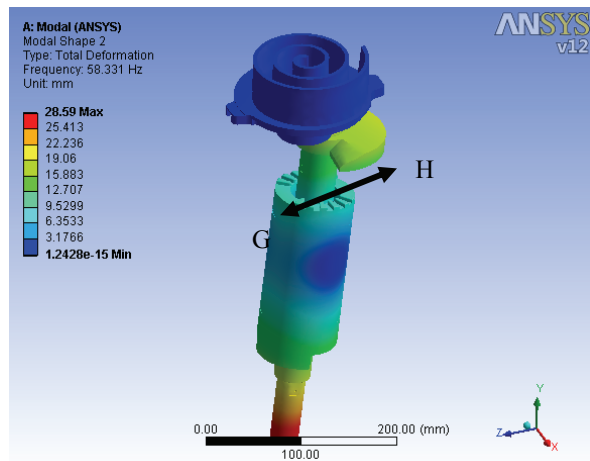


Figure 15 the 2nd modal shape for assemble parts

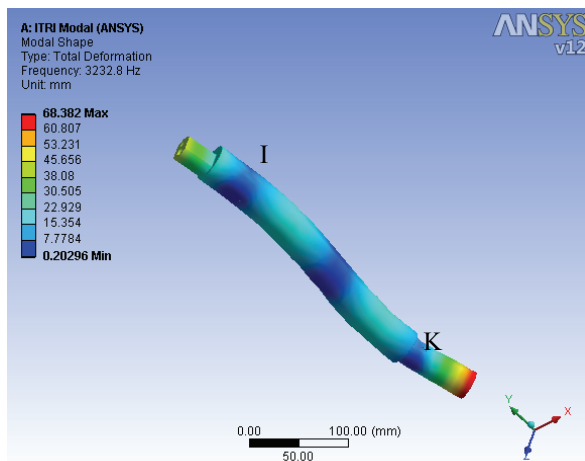


Figure 16 the modal shape of shaft

6. DISCUSSION AND CONCLUSION

By giving the inlet and outlet temperature of compressor, the temperature distributions for fixed and orbiting scroll were shown in Figure 2 and 3, respectively. The temperature distribution gradually changed. The thermal stresses were important for the compressor stress analysis.

The total deformations with temperature effects for fixed and orbiting scroll were shown in Figure 4 and 5, respectively. The capital letter A area was the maximum deformation area. The maximum deformation was 0.046mm for fixed scroll and 0.055mm for orbiting scroll. The material choice was important here. It showed that the orbiting scroll should be stiffer than the fixed scroll to avoid the two scrolls interfere with each other.

In the Figure 7, it showed the polished area, says letter B, for fixed scroll. From the stress analysis for fixed scroll, it showed the identical area for maximum stress area in Figure 6. The letter B area was not from temperature but pressure only in Figure 8. The main effect on the polished area was temperature effect. That made the structural weaker at that area. After the analysis, the designer had to make structural modifications at the weaker part.

In the Figure 9~10, the letter C on stress distribution for orbiting scroll showed the identical polished area for orbiting scroll in Figure 11.

In the Figure 12 and 13, the letter D on stress distribution for orbiting scroll was located at identical location.

The first resonance of assemble parts happened at 3236.94 rpm in Figure 14. The second resonance of assemble parts happened at 3499.86rpm in Figure 15. The compressor speed should not stay those two values in order to avoid the resonance.

In the Figure 16, the picture of the maximum deformation for modal shape showed the nodal point for the shaft. These locations were the best positions for bearing. (Letter I and K) The designer didn't need to have a huge roller bearing at those nodal points.

The effective dimensions for every component shown here were classified to protect the intellectual property rights. The simulated results showed the very good trend compared with the empirical results. With those results, it surely predicted the empirical results without making several real models. In the simulation software, it was very easy to change the model dimension. The simulation results decided a better model for production. The results answered the best positions for bearing; the resonant speed of scroll compressor; the weakest part of components. The designer can improve the prototype according to that information from mechanics view without producing several real new models. The time and cost saving for compressor production were huge.

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